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RESEARCH MEMORANDUM

COMPARISON OF EXPERIMENTALLY AND ANALYTICALLY DETERMINED WINDMILLING CHARACTERISTICS OF A COMPRESSOR WITH LOW OVER-ALL PRESSURE RATIO

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COMPARISON OF EXPERIMENTALLY AND ANALYTICALLY DETERMINED WINDMILLING CHARACTERISTICS OF A COMPRESSOR WITH LOW OVER-ALL PRESSURE RATIO

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SUMMARY

Windmilling tests on a two-stage transonic compressor with a design over-all pressure ratio of 1.85 produced the following results. The maximum windmilling equivalent speed was 30.2 percent of design speed, and the maximum windmilling equivalent flow was 57.4 percent of design flow. The maximum flow was limited by choking in the last stator passage. A method was developed for estimating maximum windmilling speed and flow from known compressor design values and geometry for a compressor with exit stators. The method, which is based on the results of the analysis of the windmilling data, predicts the maximum windmilling equivalent speed within 3.3 percent and the maximum windmilling equivalent flow within 0.4 percent.

INTRODUCTION

When an axial-flow compressor is used in a high flight Mach number application such as a turborocket, it may be desirable to windmill the compressor at high flight Mach numbers to obtain better fuel economy. The required compressor pressure ratio at takeoff for this application is in the neighborhood of 1.5 to 2.0. The problem is whether the compressor will pass the required mass flow under windmilling conditions. As a general rule the inlet equivalent flow at the low flight Mach number condition where the compressor is driven is higher than the equivalent flow at high flight Mach numbers when the compressor is windmilling. Therefore, it may be necessary that the compressor pass only a part of compressor design equivalent flow under windmilling conditions. It is important that the maximum windmilling flow be known, since the thrust is proportional to the flow. However, the margin between maximum and required windmilling flow must be sufficient so that the total-pressure drop across the compressor is not excessive.

In order to determine the windmilling characteristics of a transonic compressor with low over-all pressure ratio, data were obtained on a two-stage transonic compressor having an over-all pressure ratio of 1.85. Since windmilling experiments are not always possible, an analytical method of predicting maximum windmilling flow is desirable. A method based on the limited results of this experiment is developed and presented herein.

SYMBOLS

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A	flow area, sq ft
$\mathscr{A}_2$	ratio of boundary-layer blockage area to annulus area after last rotor
$\mathscr{A}_{2s}$	ratio of boundary-layer blockage area to passage area in last stator passage
a	velocity of sound, ft/sec
b	blade thickness, ft (fig. 3)
c_p	specific heat at constant pressure, ft-lb/(slug)($^{\circ}$R), equal to Jgc_p in engineering units
g	acceleration due to gravity, ft/sec^2
J	mechanical equivalent of heat, ft-lb/Btu
K	proportionality constant (eq. (2))
N	compressor speed, rpm
$N/\sqrt{\theta}$	equivalent compressor speed, rpm
n	number of blades in last stator blade row
P	total pressure, lb/sq ft
r	radius, ft
S	number of compressor stages
T	total temperature, $^{\circ}$R
U	rotor velocity, ft/sec

V gas velocity, ft/sec

w weight flow, lb/sec

$w\sqrt{\theta}/\delta$ equivalent weight flow, lb/sec

Y $\left(\frac{P_3}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1$

β air angle, angle between air velocity and axial direction, deg

γ ratio of specific heats

δ ratio of inlet total pressure to NACA standard sea-level pressure of 2116 lb/sq ft

ε angle between axial direction and tangent to pressure surface at inlet to last stator

η windmilling efficiency, $\dfrac{1 - \frac{T_3}{T_1}}{1 - \left(\frac{P_3}{P_1}\right)^{\frac{\gamma-1}{\gamma}}}$

η_2 windmilling efficiency based on total-pressure ratio at inlet of last stator, $\dfrac{1 - \frac{T_3}{T_1}}{1 - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}}$

θ ratio of inlet total temperature to NACA standard sea-level temperature of 518.7° R

ρ static density, slugs/cu ft

Subscripts:

a based on stagnation conditions

c choking condition

D design

d driven

h hub

i axial station at inlet to last rotor

m mean

s axial station at minimum area in last stator passage 4809

t tip

w windmilling

z axial direction

θ tangential direction

1 axial station at inlet to compressor

2 axial station at inlet to last stator

3 axial station at exit of compressor

Superscript:

$'$ relative to rotor blade row

APPARATUS AND PROCEDURE

The compressor tested is essentially the one described in reference 1. For the windmilling tests the compressor was uncoupled from the gearbox and the speed was varied by adjusting the compressor discharge pressure. The compressor bearings for this setup consisted of two spherically seated silver-plated journal bearings with lead-tin coating. Both bearings were 2 inches in diameter and 1.25 inches long, with 0.004-inch diametric clearance. The thrust bearings were tapered land type with a surface area of approximately 58 square inches for both the main and incidental bearings.

Inlet stagnation conditions to the compressor were measured by ten total-pressure probes and five total-temperature probes in the inlet depression tank. Exit stagnation pressure was measured by means of a 14-tube circumferential rake that covered one passage of the second stator. Readings from the rake were taken at five radial positions that corresponded to centers of five equal areas of the annulus. Exit stagnation temperature was measured by a radial rake having five spike-type iron-constantan thermocouples located at area centers of five equal annular areas. Compressor speed was determined by feeding the signal from a magnetic pickup located over the first-stage rotor blade row into an electronic counter. Weight flow was measured with an adjustable orifice.

The compressor windmilling speed was increased in steps from a very low speed to top windmilling speed by opening the compressor discharge throttle. To determine the effect of the second stator blade row on windmilling performance, the compressor was also tested with these stator blades removed.

RESULTS AND DISCUSSION

The compressor driven total-pressure ratio is plotted against inlet equivalent flow for equivalent speeds from 50 to 110 percent of design in figure 1. The peak-efficiency line indicates the total-pressure ratio at which peak efficiency occurred at each speed. The compressor design point will be considered the peak-efficiency point at design speed, where the over-all total-pressure ratio is 1.85 and the equivalent flow is 101 pounds per second.

In figure 1 are also presented the variations of windmilling total-pressure ratio, speed, and efficiency with inlet equivalent flow for the complete compressor and for the compressor with the second stator blade row removed $\left(1\frac{1}{2}\text{-stage compressor}\right)$. The maximum windmilling flow limit is approximately 58 pounds per second with both stages or 57.4 percent of design equivalent flow. The corresponding maximum windmilling equivalent speed is 3170 rpm or 30.2 percent of the design equivalent speed of 10,504 rpm. When the second-stage stator blades were removed, the speed was unchanged for a given equivalent flow; but the pressure drop was reduced to approximately half the original value, which indicates that half of the loss in total pressure occurs across the last stator. The maximum equivalent speed of 3740 rpm occurred when the annulus downstream of the second rotor choked at an equivalent flow of 64 pounds per second.

The windmilling efficiency was increased by removal of the second row of stator blades since, for a given flow, the speed and consequently the total-temperature drop remained constant and the total-pressure drop decreased. Efficiency was computed by the same method used in obtaining the efficiency of a turbine. The absolute magnitude of the efficiency is somewhat questionable because of the low pressure and temperature drops involved. At the top windmilling speed of 3803 actual rpm, the total-temperature drop was 3.2° F and the friction power loss computed from the temperature drop and flow was 42 horsepower. It should be noted that with no bearing friction there would be constant total temperature through the machine, with resulting zero windmilling efficiency.

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The peak-efficiency total-pressure-rise-ratio points $(P_3/P_1) - 1$ from the compressor map in figure 1 are plotted against the equivalent-speed ratio in figure 2. The curve of pressure-rise ratio obtained from the relation $Y = Y_D\left[\frac{N/\sqrt{\theta}}{(N/\sqrt{\theta})_D}\right]^2$ against speed ratio is also shown. The agreement between the peak-efficiency total-pressure-rise-ratio points and the curve obtained from the equation is good. This agreement is expected in a machine with low over-all total-pressure ratio, since the maximum efficiency is roughly the same at all speeds and the temperature rise varies approximately as the square of the wheel speed, as does Y.

The windmilling total-pressure-loss ratios for the two-stage and the $1\frac{1}{2}$-stage compressors $\left(\frac{P_3}{P_1} - 1 \text{ and } \frac{P_2}{P_1} - 1\text{, respectively}\right)$ are also plotted against equivalent-speed ratio in figure 2. The flow-ratio data obtained from the windmilling tests together with the three choke-flow points that could be obtained from the performance map of figure 1 are also shown. The total-pressure-loss ratio up to the inlet to the last stator is very nearly a mirror image of the curve for the driven data. The agreement in the shape of the two curves may be explained by the fact that driven total-pressure rise is a function of blade cambers and wheel speed squared. Cascade total-pressure loss is a function of blade camber and the square of the inlet velocity. Since the windmilling flow varies almost linearly with speed, the pressure loss would be expected to vary as a function of the blade camber and wheel speed squared. The rapidly increasing total-pressure drop across the second stator with increase in flow and speed indicates an approach to choking in the second stator that limits the maximum windmilling speed and flow. Similar trends in the pressure-drop ratio have been noted in unpublished windmilling test data from another two-stage compressor of lower over-all total-pressure ratio.

METHOD FOR ESTIMATING MAXIMUM FLOW AND SPEED

The preceding results suggest the possibility of estimating the maximum windmilling speed and flow for a compressor from known design parameters. The compressor is assumed to have a converging annulus, which results in a minimum flow area in the last stator blade passage. Therefore, choking at this point limits the windmilling flow and consequently limits the windmilling speed. If the minimum flow area is in the downstream annulus, an approach similar to that for the downstream stator can be used. When choking occurs in the rotor or an upstream stator, the analysis presented does not apply.

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The method for determining maximum windmilling inlet equivalent flow may be summarized as follows:

(1) Determine a choking flow for the downstream stator. To determine this value a flow area is determined from the geometry for this blade row.

(2) To determine the inlet equivalent flow, the windmilling total-pressure loss from the compressor inlet to the inlet of the last stator must be determined. The results of this experiment show that the pressure loss depends upon wheel speed.

(3) From the last-stator equivalent choking flow and the assumption that the relative air angle leaving the last rotor at the mean radius is equal to the design value, the wheel speed may be determined. Consequently, from the results of steps (1) and (2) the maximum windmilling inlet equivalent flow can be found.

Estimation of Maximum Windmilling Flow

As stated previously, the maximum windmilling flow is limited by choking in the last stator passage. Therefore, to obtain an estimate of the flow it is necessary to approximate choking area. The minimum area is assumed to be determined by a perpendicular to the pressure surface of the stator blade at the inlet. Figure 3 illustrates the velocities and blade geometry involved. Calculations are based on mean-radius conditions at the inlet to the last stator. Since

$$\left(\frac{w\sqrt{\theta_2}}{\delta_2}\right)_c = 49.45\ (1 - \mathcal{A}_{2s})\ A_{2s}$$

and

$$\left(\frac{w\sqrt{\theta_1}}{\delta_1}\right) = \left(\frac{w\sqrt{\theta_2}}{\delta_2}\right)_c \left(\frac{P_2}{P_1}\right)_w \left(\sqrt{\frac{T_1}{T_2}}\right)_w$$

$$= 49.45\ (1 - \mathcal{A}_{2s})\ A_{2s} \left(\frac{P_2}{P_1}\right)_w \left(\sqrt{\frac{T_1}{T_2}}\right)_w$$

$$= 49.45 \left(\frac{P_2}{P_1}\right)_w \left(\sqrt{\frac{T_1}{T_2}}\right)_w (1 - \mathcal{A}_{2s}) \left[\pi(r_t + r_h)_2 \cos \varepsilon_m - b_m n\right](r_t - r_h)_2 \tag{1}$$

then the maximum windmilling flow can be found if $(P_2/P_1)_w$ is known, because it may be assumed that $\sqrt{T_1/T_2} = 1$. For the maximum windmilling speed for these tests, $\sqrt{T_1/T_2} = 1.003$. For the subject compressor, $(P_2/P_1)_w$ at maximum windmilling speed was approximately 0.94; and, if no total-pressure losses are assumed, the estimated flow will be high by approximately 6 percent.

Estimation of Maximum Windmilling Pressure Ratio 4809

It may be assumed that the windmilling pressure-loss ratio is a function of the driven pressure-rise ratio at maximum efficiency at equal equivalent speeds:

$$\left(\frac{P_2}{P_1}\right)_w - 1 = K\left[\left(\frac{P_3}{P_1}\right)_d - 1\right]$$

$$= K\left\{\left[Y_D\left(\frac{N}{N_D}\right)^2 + 1\right]^{\frac{\gamma}{\gamma-1}} - 1\right\} \tag{2}$$

This assumption applies only when there is little energy extracted from the air to overcome bearing friction. If appreciable shaft power is extracted, the analysis would not be expected to hold.

Values were assigned to K of -1.00, -0.75, -0.50, and 0, and the results obtained from equation (2) were plotted in figure 4 together with the measured windmilling pressure-loss ratio with the second stator removed. The curve for K = -1.00 agrees well with the windmilling loss ratio for $1\frac{1}{2}$ stages up to the maximum windmilling speed for both stages.

Estimation of Maximum Windmilling Speed

The maximum windmilling speed must be estimated in order to determine the windmilling pressure ratio from equation (2) to be used in equation (1) to obtain the maximum windmilling flow. If it is assumed that under windmilling conditions $\beta'_{2,m}$ is equal to the design value, it may be noted from figure 3 that

$$U_{2,m} = V_{2,m} \cos \beta_{2,m} \tan \beta'_{2,D,m} + V_{\theta,2,m} \tag{3}$$

Since the energy removal per stage to overcome bearing friction is small, the required turning through the rotor is small. Therefore,

$$\beta_{2,m} \approx \beta_{1,m} = \beta_{1,D,m}$$

and

$$V_{z,2} \approx V_2 \cos \beta_{1,D,m}$$

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Then, from equation (3),

$$U_{2,m} \approx V_{2,m} \cos \beta_{1,D,m} \tan \beta'_{2,D,m} + V_{\theta,2,m} \tag{4}$$

From Euler's turbine equation, the expression for windmilling efficiency, and the assumption that the energy removal is equally divided between stages,

CW-2

$$V_{\theta,1,m} - V_{\theta,2,m} \approx \frac{c_p(T_1 - T_2)}{S\left(\frac{U_{1,m} + U_{2,m}}{2}\right)} = \frac{c_p T_1 \eta_2 \left[1 - \left(\frac{P_2}{P_1}\right)_w^{\frac{\gamma-1}{\gamma}}\right]}{S\left(\frac{U_{1,m} + U_{2,m}}{2}\right)} \tag{5}$$

The bearing friction horsepower, and consequently the compressor total-temperature drop, is proportional to the square of the wheel speed; the windmilling total-pressure drop has shown the same trend. Therefore, windmilling efficiency should be fairly constant with speed, as verified by figure 1. Substituting $(P_2/P_1)_w$ from equation (2) into equation (5) and substituting the result into equation (4) for $V_{\theta,2,m}$ give

$$\left(\frac{N}{N_D}\right)^2 \frac{r_{m,2}}{r_t} = \frac{N}{N_D} \frac{\cos \beta_{1,D,m}}{U_{D,t}} (V_{1,m} \tan \beta_{1,D,m} + V_{2,m} \tan \beta'_{2,D,m})$$

$$- \frac{c_p T_1 \eta_2}{S U_{D,t}^2 \left(\frac{r_{1,m} + r_{2,m}}{2r_t}\right)} \left[1 - \left(K\left\{\left[Y_D\left(\frac{N}{N_D}\right)^2 + 1\right]^{\frac{\gamma}{\gamma-1}} - 1\right\} + 1\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{6}$$

where η_2 is an assumed windmilling efficiency up to the last stator, and S is the number of stages. In the subject compressor, $\beta_{1,D,m}$ is zero, and η_2 is approximately 0.25.

The two velocities $V_{1,m}$ and $V_{2,m}$ must be evaluated, and values for K and η_2 must be assumed before equation (6) may be solved by trial and error for N/N_D. Equation (1) and conservation of mass may be used to derive the following for evaluating $V_{2,m}$ for the stator choke condition:

$$\left(\frac{\rho V}{\rho_a a_a}\right)_{2,m} = \frac{\dfrac{w\sqrt{\theta_1}}{\delta_1}}{g(\rho_a a_a)_2\ (1 - \mathscr{A}_2)A_2 \cos \beta_{2,m}}$$

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$$= \frac{\dfrac{w\sqrt{\theta_2}}{\delta_2}}{g(\rho_a a_a)_1\ (1 - \mathscr{A}_2)A_2 \cos \beta_{2,m}}$$

$$= \frac{49.45(1 - \mathscr{A}_{2s})A_{2s}}{g(\rho_a a_a)_1\ (1 - \mathscr{A}_2)A_2 \cos \beta_{2,m}}$$

or

$$\left(\frac{\rho V}{\rho_a a_a}\right)_{2,m} \approx \frac{0.58(1 - \mathscr{A}_{2s})A_{2s}}{(1 - \mathscr{A}_2)A_2 \cos \beta_{i,D,m}}$$

$$= 0.58\ \frac{1 - \mathscr{A}_{2s}}{1 - \mathscr{A}_2}\ \frac{\pi(r_t + r_h)_2 \cos \varepsilon_m - b_m n}{\pi(r_t + r_h)_2 \cos \beta_{i,D,m}} \tag{7}$$

From equation (7) a unique choking velocity is determined in the annulus ahead of the last stator, since $(\rho/\rho_a)_{2,m}\ (V/a_a)_{2,m}$ against $(V/a_a)_{2,m}$ may be determined from reference 2, and then $V_{2,m}$ may be evaluated if $\sqrt{T_2/T_1}$ is assumed to be unity $(a_{a,2} = a_{a,1})$. Equation (7) may be used for approximating $V_{1,m}$ by substituting $(1 - \mathscr{A}_1)A_1$ for $(1 - \mathscr{A}_2)A_2$. This assumes that $(P_2/P_1)\sqrt{T_1/T_2}$ is unity.

After equation (6) is solved for N/N_D, equation (2) may be used to obtain $(P_2/P_1)_w$, and this value may be used in equation (1) to obtain the maximum windmilling inlet equivalent flow.

Results of Estimations

The estimated maximum windmilling speed and flow for the subject compressor were determined by the preceding method with the assumptions that $\mathscr{A}_2 = 0.04$ (design value), $\mathscr{A}_{2s} = 0.08$, $\eta_2 = 0.25$, and $K = -1.00$. The results were $N/N_D = 0.312$ and $(w\sqrt{\theta_1}/\delta_1)/(w\sqrt{\theta_1}/\delta_1)_D = 0.572$. The predicted maximum speed and flow are within 3.3 percent and 0.4 percent, respectively, of the experimentally determined values.

In the following table are presented the errors in predicted maximum windmilling speed and flow due to a variation of the necessary assumptions. The percentage error is based on the experimentally determined values of maximum equivalent speed and equivalent flow of 3170 rpm and 58 pounds per second, respectively. The first horizontal row represents the values quoted previously:

K	η_2	$\mathscr{A}_2$	$\mathscr{A}_{2s}$	Equivalent speed, rpm	Speed error, percent	Equivalent flow, lb/sec	Flow error, percent
-1.00	0.25	0.04	0.08	3275	3.3	57.8	-0.4
0	.25	.04	.08	3610	13.9	62.1	7.1
-1.00	0	.04	.08	3610	13.9	57.0	-1.7
-1.00	.25	0	0	3465	9.3	62.4	7.6
0	0	0	0	3830	20.8	67.5	16.4

Quite large variations in the assumed values have little effect on the predicted flow. For example, assuming no bearing friction ($\eta_2 = 0$) results in a fairly high percentage error in predicted speed, but the error in flow is less than 2 percent. Errors in the stator boundary-layer blockage factor ($\mathscr{A}_{2s}$) affect the predicted flow directly.

For a compressor with anti-friction bearings, the total-temperature and total-pressure drops would be somewhat lower than with journal bearings, but it is felt that the effect on the maximum windmilling flow would be slight.

SUMMARY OF RESULTS

Windmilling tests on the two-stage transonic compressor yielded the following results:

1. The maximum windmilling equivalent speed was 30.2 percent of design speed, and the maximum windmilling equivalent flow was 57.4 percent of design flow.

2. Choking in the second-stator passage limited the maximum flow.

3. For this compressor the windmilling pressure-drop ratio up to the inlet to the last stator blade row was approximately numerically equal to the driven maximum-efficiency pressure-rise ratio at corresponding percentages of equivalent-design speed. 4809

4. A method developed for approximating the maximum windmilling speed and flow by use of known compressor design values and geometry predicted the maximum windmilling equivalent speed within 3.3 percent and the maximum windmilling equivalent flow within 0.4 percent.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, December 18, 1957

REFERENCES

1. Hatch, James E., and Bernatowicz, Daniel T.: Aerodynamic Design and Over-All Performance of First Spool of a 24-Inch Two-Spool Transonic Compressor. NACA RM E56L07a, 1957.

2. The Staff of the Ames 1- by 3-Foot Supersonic Wind-Tunnel Section: Notes and Tables for Use in the Analysis of Supersonic Flow. NACA TN 1428, 1947.

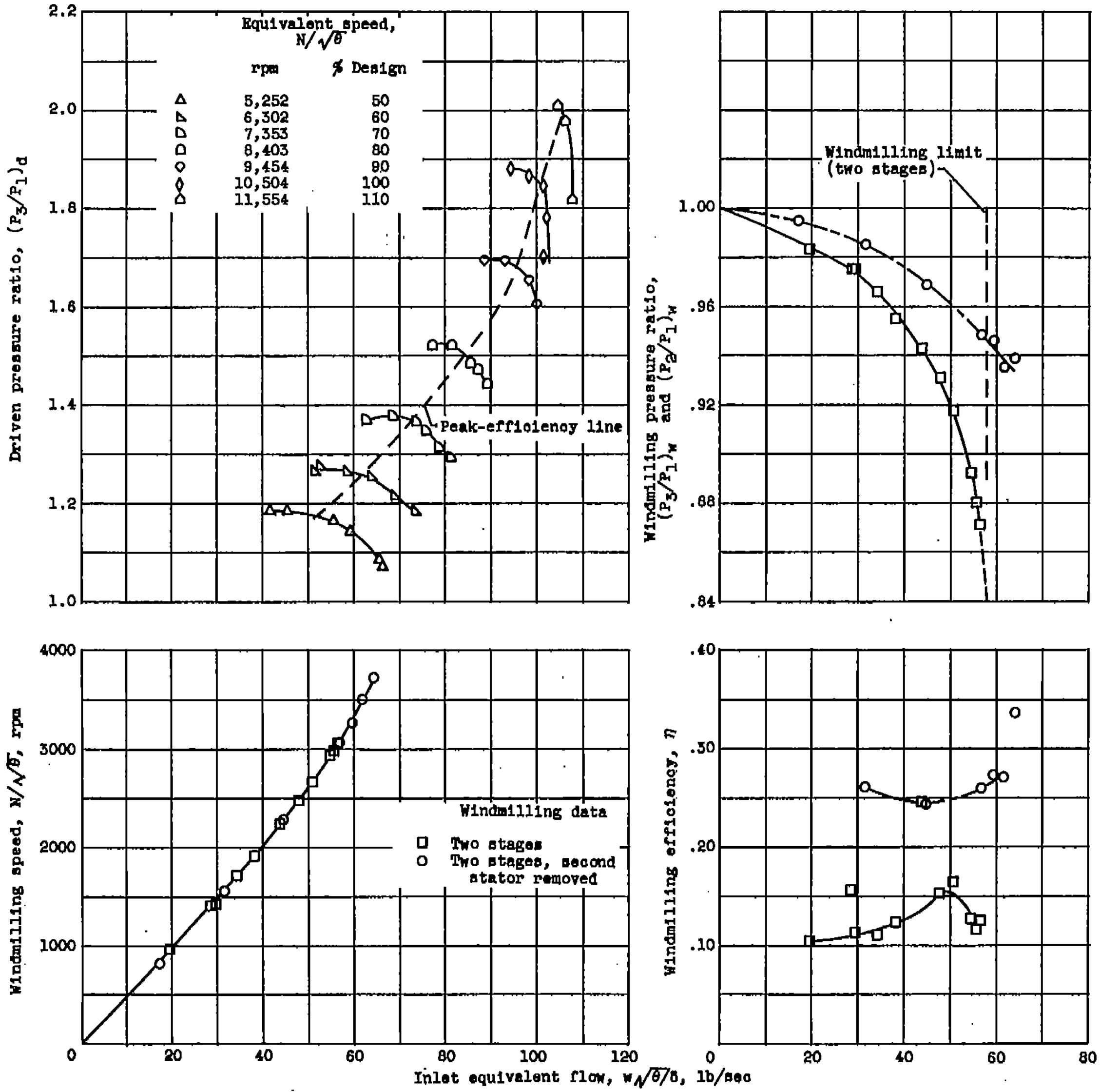

Figure 1. - Variation of windmilling and driven performance with inlet equivalent flow.

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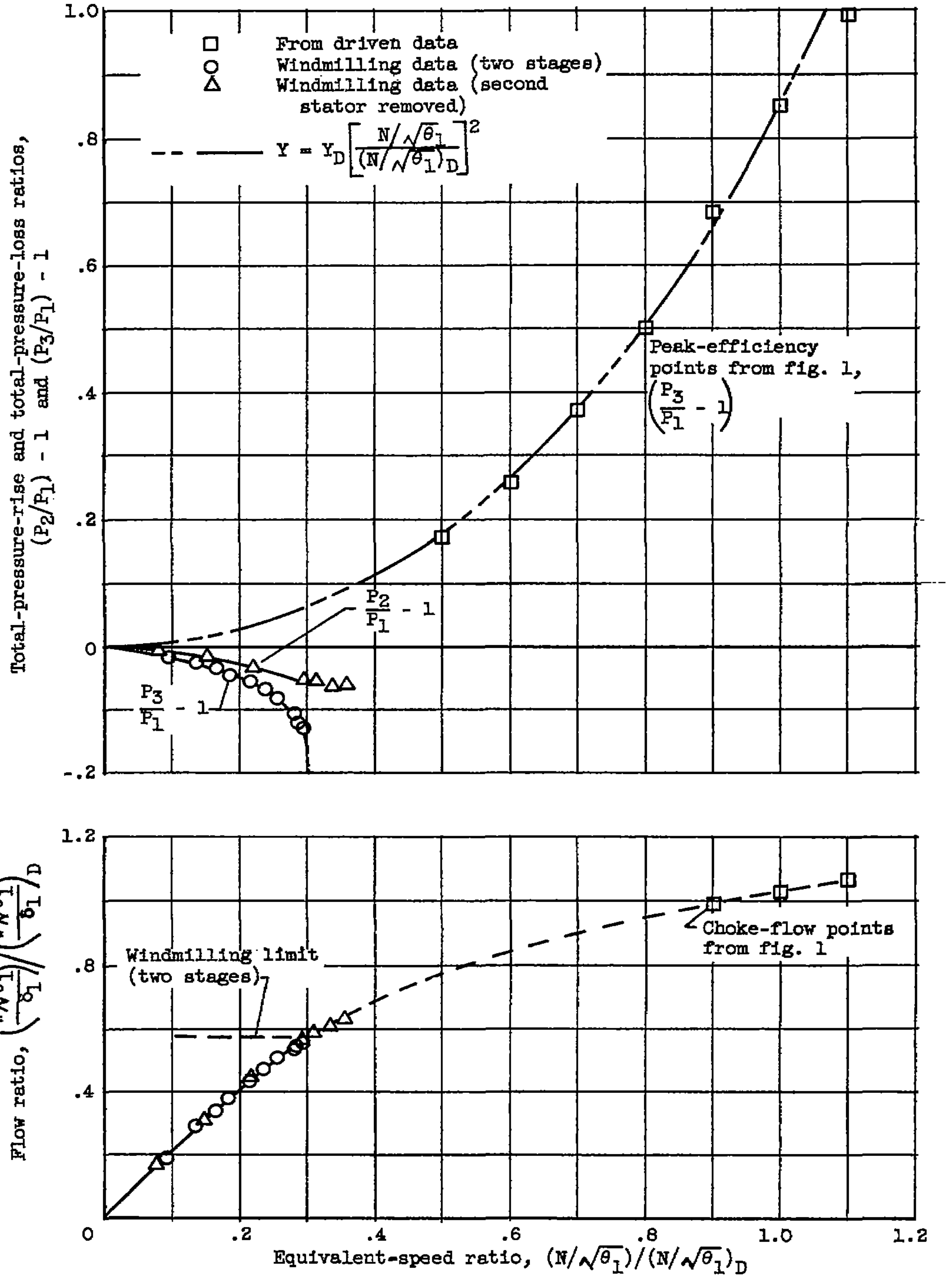

Figure 2. - Variation of flow ratio and pressure-rise and -loss ratios with speed ratio.

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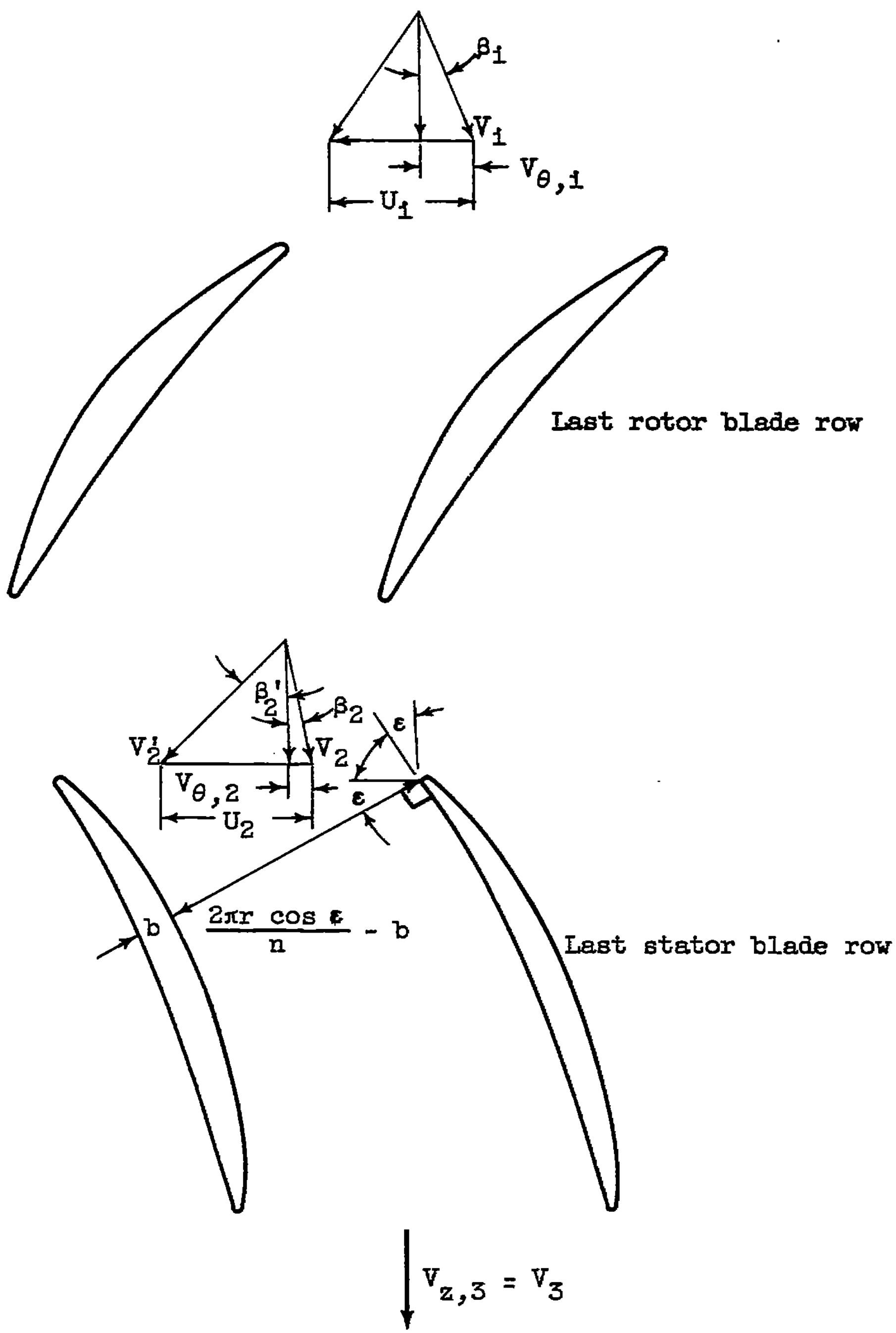

Figure 3. - Velocity diagrams and geometry at mean radius of last stage.

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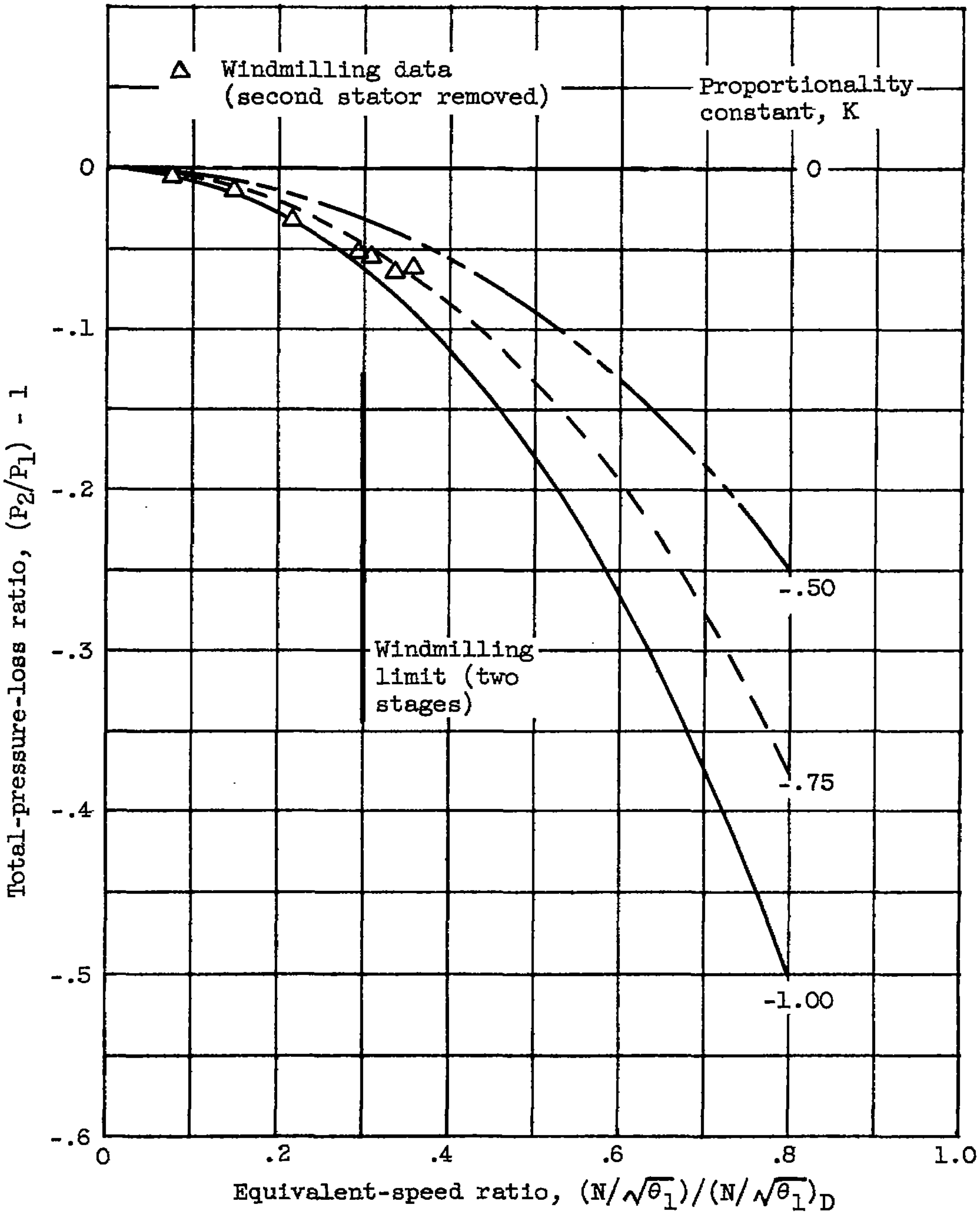

Figure 4. - Comparison of measured and calculated total-pressure loss up to second stator.

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